An Investigation of a Series of Radial Blade Centrifugal Fans

A THESIS

Presented to The Mechanical Engineering Department Of The Georgia School of Technology in Partial Fulfillment of the Requirements for the Degree of MASTER OF SCIENCE

By

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INTRODUCTION

Object

This investigation was undertaken for the purpose of establishing performance charts and tables for a series of six radial blade centrifugal fans. It was necessary that the tables and charts be complete enough to allow the manufacturer to select the proper size fan and the speed at which it should be driven for any conditions of capacity and pressure within the possible range of performance of the machines.

Acknowledgment

The test contained in Chapter II was made in the Bessemer, Alabama, Works of the United States Pipe and Foundry Company.

The author is indebted to Mr. L. W. Williams of the above company for his kind permission to use the test data.

Theoretical Considerations

Head Produced by a Fan. From a consideration of the energies involved at sections on the suction and discharge sides of a fan, the following equation results:

Suction Side			Discharge Side		
Static Head	Velocity Head	Input Energy	Static Head	Velocity Head	Friction Loss

$$W\left(p_1 + z_1 + \frac{v_1^2}{2g}\right) + WH = W\left(p_2 + z_2 + \frac{v_2^2}{2g}\right) + W_f \qquad (1)$$

where

W = weight of air flowing per second;

P_1 = static pressure above atmosphere at section 1, ft. of air;

z_1 = vertical height of section 1 with reference to some datum plane;

v_1 = average velocity at section 1;

g = acceleration due to gravity;

H = total head on fan;

P_2 = static pressure above atmosphere at section 2;

z_2 = vertical height of section 2 with reference to same datum plane as for section 1;

v_2 = average velocity at section 2;

f = loss in pressure between the sections due to friction.

Rearranging and dividing through by W, equation 1 may be written as

$$H = (p_2 - p_1) + (z_2 - z_1) + \left(\frac{v_2^2 - v_1^2}{2g}\right) + f. \quad (2)$$

The term $(z_2 - z_1)$ is small in comparison with the other terms of equation (2) and is therefore neglected. If the section 1 is taken in the atmosphere on the suction side of the fan p_1 is the atmospheric or barometric pressure and v_1 is zero.

Thus

$$H = p_2 + \frac{v_2}{2g} + f \quad (3)$$

becomes the total head, in feet of air, against which the

the fan operates. It should be noticed that p_2 is simply the static pressure, above the atmospheric pressure, in the discharge duct, expressed in feet of air.

Air Horsepower. It is necessary to know how much power is contained in a moving column of air at the discharge of the fan, in order to determine the mechanical efficiency of the fan. Thus by definition

$$\text{Horsepower} = \frac{\text{Foot lb. per min.}}{33000}$$

$$= \frac{\text{Weight of Air per min.} \times H}{33000}$$

$$= \frac{CFM \times w \times H}{33000}$$

where

CFM = cubic feet of air flowing per minute

w = density of air in pounds per cubic foot

H = total head in feet of air (equation 3)

expressing H in inches of water

$$H = \frac{TP}{12} \times \frac{62.3}{w}$$

where TP = total head on discharge side expressed in inches of water

62.3 = density of water in pounds per cubic foot

therefore

$$\text{Horsepower} = \frac{CFM \times w \times TP \times 62.3}{33000 \times 12 \times w}$$

or

$$\text{Air Horsepower} = \frac{CFM \times TP}{6356} \qquad (4)$$

Mechanical Efficiency of a Fan. The mechanical efficiency of a fan is defined as the ratio of the air horsepower to the brake horsepower supplied to the fan shaft.

Determination of Air Velocity in a Duct. It is necessary to know the average velocity in the discharge duct, in order to compute the air horsepower and the volume pumped. It is recognized that a correctly built pitot tube will give the velocity at a point according to the following law;

$$v = \sqrt{2gh}$$

where v = velocity in feet per second

g = acceleration due to gravity

h = velocity pressure, i. e., the difference between the impact and static pressures as given by the pitot tube, in feet of air.

expressing h in inches of water

$$h = \frac{VP}{12} \times \frac{62.3}{w}$$

where w = air density in pounds per cubic foot

VP = velocity pressure in inches of water

substituting in the above formula and expressing the velocity in feet per minute

$$V = 60\ v = 60\sqrt{2g\left(\frac{VP \times 62.3}{12 \times w}\right)}$$

$$= 1096.2\sqrt{\frac{VP}{w}} \qquad (5)$$

It should be noticed that if V is the average velocity in the duct, that it is necessary for VP to be the average velocity pressure in the duct. Also, w is the density of the air in the duct.

Determination of the Average Velocity. Twenty complete sets of readings are taken with the pitot tube along two traverses ninety degrees apart. The duct is divided into five concentric areas as shown in Figure 1. The tube is placed at the center of each of these areas and four readings are taken. The points at which the tube is to be placed are found in the following manner:
let r_1, r_2, r_3, r_4, r_5, equal the respective radius of each of the concentric areas; let R equal the radius of the duct; now since

$$\pi r_1^{\;2} = \pi R^2 \times .2\;, \quad r_1 = \sqrt{.2}\;R$$

Similarly

$$r_2 = \sqrt{.4}\;R$$

$$r_3 = \sqrt{.6}\;R$$

$$r_4 = \sqrt{.8}\;R$$

Now the center of area of each of these areas will be on a line which divides each of the small areas into two equal areas, thus

$$\text{point 1, } \frac{\pi r_1^{\;2}}{2} = \pi R^2 \times \frac{.2}{2} = \pi R^2 \times .1, \; r_1' = \sqrt{.1}\;R$$

$$\text{or } r_1' = .316\;R$$

Similarly

$$r_2' = \sqrt{.3}\;R = .548\;R$$

$$r_3' = \sqrt{.5}\;R = .707\;R$$

$$r_4' = \sqrt{.7}\;R = .837\;R$$

$$r_5' = \sqrt{.9}\;R = .949\;R$$

It is evident that the average velocity will be

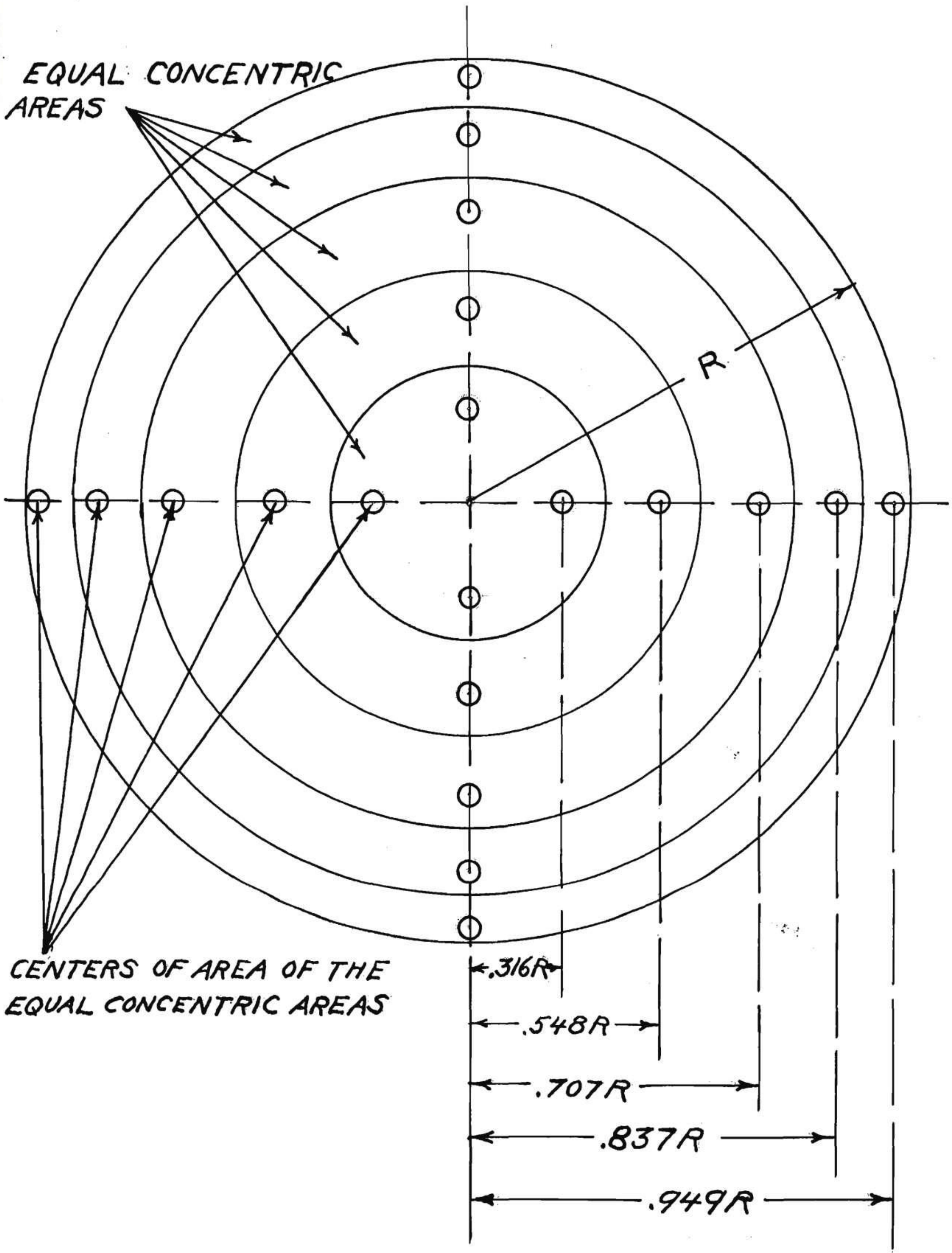

EQUAL CONCENTRIC AREAS
R
CENTERS OF AREA OF THE EQUAL CONCENTRIC AREAS
.316R
.548R
.707R
.837R
.949R

FIGURE 1

the average of the velocities corresponding to the various pressure readings. Thus the average velocity pressure will not be the average of the pressure readings but will be the pressure corresponding to the average velocity. Since the velocity varies as the square root of the velocity pressure, the average velocity pressure will be given by

$$VP = \left(\frac{\sqrt{a} + \sqrt{b} + \sqrt{c}\ .\ .\ .\ etc.}{n} \right)^2 \qquad (6)$$

where

VP = average velocity pressure

a, b, c, etc. = various velocity pressures at each point of the traverse

n = number of pitot tube readings in each traverse.

The average velocity will be secured if the velocity pressure as defined by equation (6) is substituted in equation (5).

Capacity. The capacity of the fan will be secured if the average velocity as determined by equation (5) is multiplied by the area of the discharge duct at the point of measurement.

Loss of Pressure due to Friction. It is usual to compute the loss in pressure due to friction with the hydraulic formula

Loss of head due to friction = $K\ P/A\ lv^2$

where

K = an experimental constant = .0001

P = perimeter of duct

A = area of duct

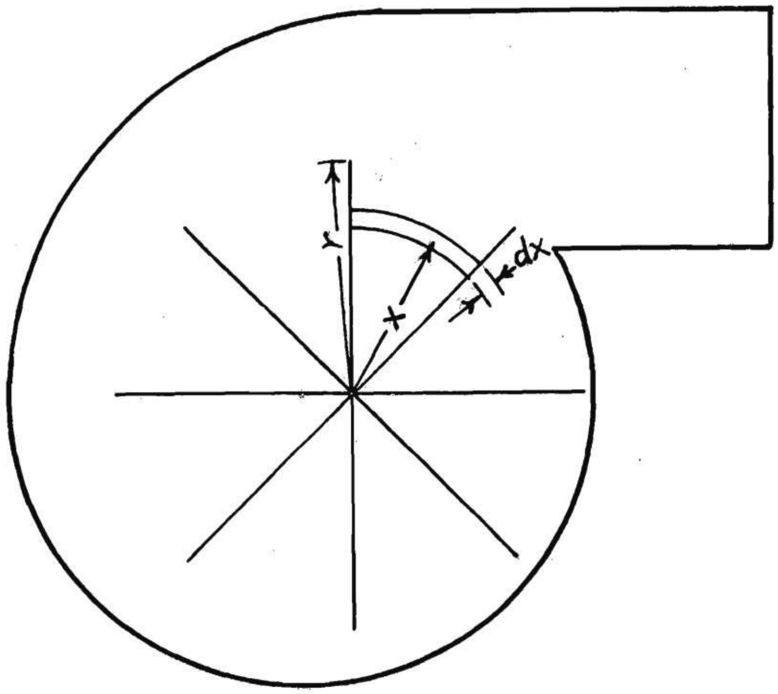


FIGURE 2

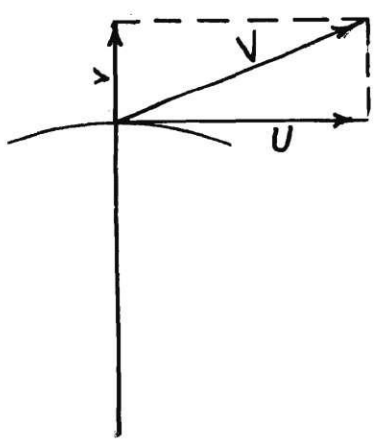


FIGURE 3

l = length of duct in feet

v = velocity of air in duct in ft. per sec.

Substituting 2gVP for v^2 and expressing P and A in terms of the diameter of the duct D, the above formula becomes

$$\text{Loss of head due to friction} = .0001 \frac{\pi D}{\frac{\pi D^2}{4}} L\ 2gVP$$

$$= .0257\ L/D\ VP \qquad (7)$$

It should be noticed that if VP is the average velocity pressure expressed in inches of water that the loss of head due to friction will also be expressed in inches of water. The pressure loss as given by equation (7) becomes the last term in equation (3).

Theory of the Centrifugal Fan. The radial blade centrifugal fan consists essentially of a number of radial blades mounted on a shaft which may be rotated in a suitable casing. Figure 2 shows the essential features of the fan.

The flow of air through the fan is due to the centrifugal force imparted to the air enclosed between the blades and also to the tangential velocity imparted to the air by the blade tips. Figure 3 gives a vector diagram of the velocities at the tip of the blade. The peripheral velocity of the blade is denoted by U, the velocity of the air relative to the blade is denoted by v, and the absolute velocity of the air leaving the blade is denoted by V. The fan housing is given a scroll shape of gradually increasing area in order to partially convert the velocity head as represented by V into useful static pressure at the fan outlet.

Referring to Figure 2, let dx be the thickness of a thin layer of air between two of the blades at a distance x from the axis and having an area of S. The volume of this layer is Sdx, and its weight will be SdxD where D is the air density. If the impeller is rotating with a constant angular velocity of w radians per second and it is assumed that the air rotates with the impeller, i. e., for zero air delivery, the thin layer of air will exibit a centrifugal force of

$$dF = \frac{SdxD}{g} w^2 x \qquad (\text{Centrifugal force} = M w^2 r)$$

The unit pressure developed

$$dP = dF/S$$

expressed in terms of feet of air head

$$dH = dP/D = dF/SD$$

$$= \frac{w^2 xdx}{g}$$

Integrating from o to r where r is the radius of the blade

$$H = \int_0^r \frac{w^2 xdx}{g} = \frac{w^2 r^2}{2g}$$

Now wr is equal to U, the peripheral velocity of the blade, hence

$$H = U^2/2g$$

Actually the head developed will be

$$H = k U^2/2g \qquad (8)$$

where

k = a constant dependent upon the fan construction.

It is evident that when the outlet is opened that the head given by equation (8) would cause a flow of air through the fan equal to $k v^2/2g$, if the air flow was totally unresisted. Therefore it may be stated that

$$v = e\ U \tag{9}$$

where

e = a constant which depends upon the amount of the outlet opening.

Thus, the capacity of the fan will be directly proportional to the peripheral speed (for a given outlet opening) since the capacity is equal to the product of the radial velocity v, and the area of the impeller at the periphery.

Now, the air horsepower is proportional to the capacity and head produced (equation 4), and since the capacity varies directly as the speed and the head varies as the square of the speed, the horsepower will vary as the cube of the speed.

Thus it may be stated that for a given size fan, operating on a given air density and at a definite per cent of outlet opening:

1. The capacity in cubic feet per minute varies directly as the fan speed.
2. The pressure developed varies as the square of the speed.
3. The horsepower required to drive the fan varies as the cube of the fan speed.

Effect of Fan Size on Speed, Capacity, Pressure Developed, Horsepower, and Efficiency. It is customary to design fans such that the various dimensions for the different sizes are all changed in proportion. Thus, if the diameter of the impeller is increased, the width of the impeller would also be increased in the same proportion, the diameter of the inlet and outlet connections would be increased, etc.

Equation (8) shows that the head developed by a fan is proportional to the square of the peripheral velocity. Thus, for a different size fan, the same head would be produced as long as the peripheral speed was the same. Therefore, since the peripheral speed varies directly as the fan size, the rotative speed would have to vary inversely as the fan size in order to secure the same peripheral speed and head.

Equation (9) pointed out the fact that the radial velocity of the air through the fan was proportional to the peripheral velocity of the blade. Thus for a given outlet opening the capacity is equal to the radial velocity of the air times the peripheral area of the impeller. Since both the impeller diameter and width are increased in proportion, the peripheral area of the impeller will vary as the square of the fan size. Therefore, the capacity of a different size fan will vary as the square of the fan size where the fan is operated at the same peripheral velocity and the same amount of outlet opening as the fan for which the capacity is known.

It has been shown that the pressure developed by a fan is proportional to the peripheral velocity of the impeller. Therefore all the fans in the series will develope the same pressure as long as the peripheral velocity is kept the same.

The fan horsepower has been shown to be dependent on the product of the capacity and the pressure developed. It has just been stated that for a different size fan that the capacity will vary as the square of the fan size and the pressure will be the same as long as the peripheral velocity is the same. Therefore, the horsepower required to drive the fan will vary as the square of the fan size.

The mechanical efficiency of different size fans which are symetrical will be the same for the same operating conditions. This is due to the fact that the area of the frictional surfaces and the volume pumped both increase in the same proportion, i.e., as the square of the fan size.

Thus, the following laws may be stated which apply to different sizes of a series of symetrical fans which are operating against the same pressure and the same amount of outlet opening:

1. The r.p.m. varies inversely as the fan size.
2. The capacity varies as the square of the fan size.
3. The horsepower varies as the square of the fan size.
4. The mechanical efficiency remains constant.

Effect of Air Density. The terms of equation (3) are expressed as feet of air pressure. However, it is customary to express the head of a fan in terms of inches of water. Thus it will be noticed that the equations for the various pressures, capacity, and horsepower are all worked out in terms of inches of water pressure. Now, it has been shown that the head produced by a fan, expressed in feet of air, depends on the peripheral velocity of the blades only, i.e., for a given velocity there will be a definite head, regardless of the air density. However, when the head is expressed in inches of water it is evident that the head will be directly proportional to the air density. The performance of fans is usually given in terms of some standard air density whereas the test of a fan is seldom carried out with air of a density corresponding to the density of standard air. Therefore, in order to express the performance of a fan in terms of standard air, it is necessary to multiply the various pressures and horsepower by the ratio of the density of standard air to that of the test air.

Method of Investigation of a Series of Symetrical Fans. The laws stated on pages 11 and 13 point out the way in which a series of symetrical fans may be investigated. Thus, if the performance of one size fan for a definite constant speed is known, the performance of this fan at other speeds may be calculated according to the laws on page 11, and the performance for the other size fans may be calculated according to the laws on page 13.

II. FAN TEST

Object

The object of the test was to determine the operating characteristics for a given size fan from no air delivery to wide open air delivery when operating at constant speed on air of standard density.

Fan Tested

Figure 4 shows the impeller and shaft assembly, and Figure 5 shows the complete fan which was tested. The data for this machine are:

Size No. 9
Impeller Diameter 40 in.
Impeller Width 15 in. maximum
Inlet and Outlet Diameters 22 in.

Arrangement of Equipment

The fan was driven by a direct current motor arranged for V-Belt drive. The information pertaining to the drive is as follows:

6 Belt, V-Belt drive
Made by Manhattan Rubber Co.
Belt No. B-128
Sheave size motor end, 14 in. pitch diameter
Sheave size fan end, 12-21/32 in. pitch diameter
Center to center distance 43 in.

A welded sheet iron duct 22 in. in diameter and ten diameters long was connected to the discharge flange of the fan. Holes were drilled in this duct at a distance of seven and one-half diameters from the outlet flange. These holes were drilled in the same plane but ninety degrees apart so as

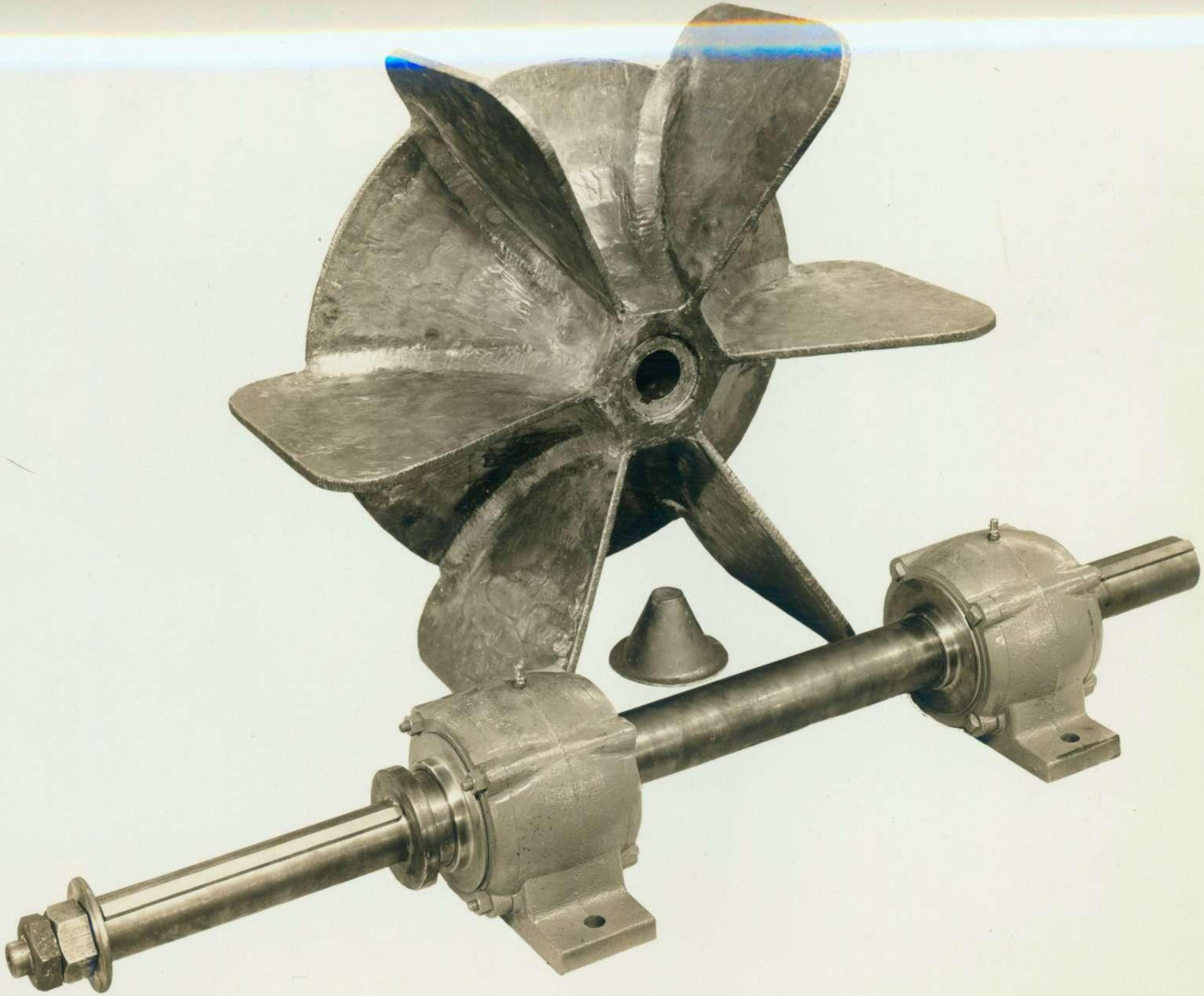


FIGURE 4

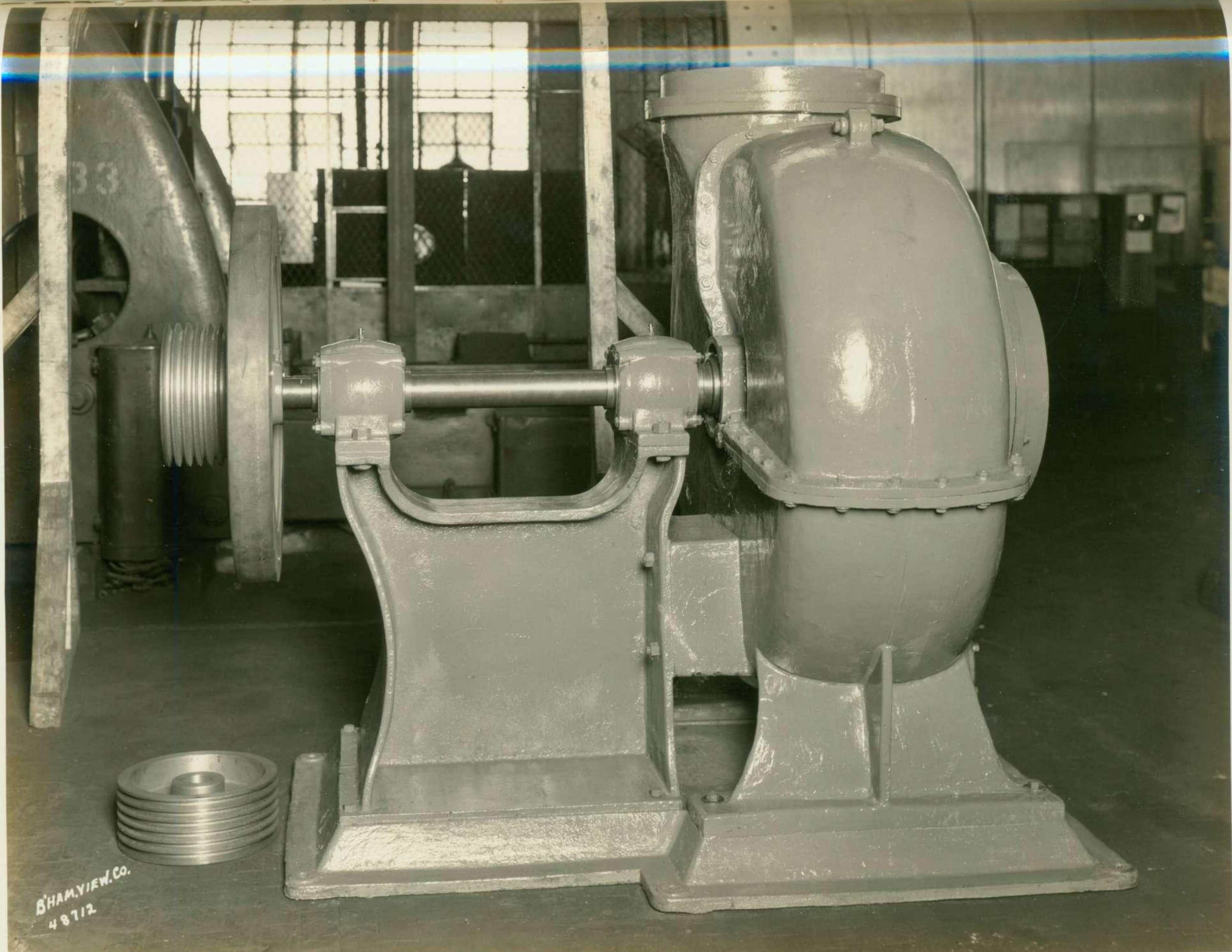


FIGURE 5.

to permit the Pitot Tube to traverse the duct in two directions. A sheet iron duct six diameters long and 22 in. in diameter was connected to the fan suction in order to duplicate conditions under which the fan is installed.

A direct current ammeter and a direct current voltmeter were connected into the power line just ahead of the motor controller.

Apparatus

The following pieces of apparatus were used in test:

Weston D.C. Voltmeter; Weston D.C. Ammeter; Wheatstone Bridge; Stop Watch; Revolution Counter; American Blower Company Standard Pitot Tube; Hays *Inclined Draft Gage*, 0-3 in. scale; Vertical Water Manometer; Vertical Oil Manometer; Mercury Barometer; Standard Mercury Thermometer.

Method of Test

The motor controller was set on one speed and kept on this same setting throughout the test. Eight separate runs were made from no air delivery to full air delivery. The amount of air delivery was controlled by throttling the end of the discharge duct with a metal cone. During each run readings of voltage, amperes input, motor speed, and fan speed were made. At the same time twenty readings of impact, velocity, and static pressure were taken with the Pitot Tube at points indicated on Figure 1. Also, during each run, the dry bulb, wet bulb, and room temperatures were taken. The barometric pressure was read once each run.

TEST DATA

PITOT TUBE READINGS

POSITION OF TUBE	RUN 1 NO AIR DELIVERY			RUN 2			RUN 3			RUN 4			RUN 5			RUN 6			RUN 7			RUN 8 WIDE OPEN		
	PRESSURE			PRESSURE			PRESSURE			PRESSURE			PRESSURE			PRESSURE			PRESSURE			PRESSURE		
	Impact	Velocity	Static	Impact	Velocity	Static	Impact	Velocity	Static	Impact	Velocity	Static	Impact	Velocity	Static	Impact	Velocity	Static	Impact	Velocity	Static	Impact	Velocity	Static
1				2.96	.07	2.89	2.54	.20	2.34	2.00	.32	1.68	1.68	.53	1.15	1.38	.63	.75	1.20	.72	.48	.99	.77	.22
2				2.97	.08	2.89	2.56	.21	2.35	2.10	.36	1.74	1.68	.56	1.12	1.40	.72	.68	1.26	.81	.45	1.02	.86	.16
3				2.97	.09	2.88	2.58	.23	2.35	2.10	.40	1.70	1.68	.60	1.08	1.44	.75	.69	1.30	.83	.47	1.10	.93	.17
4				2.97	.095	2.88	2.58	.24	2.34	2.10	.41	1.69	1.72	.62	1.10	1.44	.75	.69	1.30	.85	.45	1.10	.93	.17
5				2.97	.10	2.87	2.58	.26	2.32	2.10	.43	1.67	1.72	.64	1.08	1.44	.74	.70	1.26	.83	.43	1.05	.90	.15
6				2.97	.09	2.88	2.58	.285	2.30	2.10	.45	1.65	1.72	.61	1.11	1.38	.66	.72	1.16	.73	.43	.96	.82	.14
7				2.97	.085	2.87	2.58	.28	2.30	2.14	.47	1.67	1.65	.58	1.07	1.34	.64	.70	1.10	.70	.40	.91	.78	.13
8				2.97	.08	2.87	2.58	.27	2.31	2.10	.46	1.64	1.64	.55	1.09	1.30	.60	.70	1.10	.69	.41	.91	.77	.14
9				2.97	.075	2.88	2.57	.25	2.32	2.10	.41	1.69	1.59	.50	1.09	1.28	.58	.70	1.06	.65	.41	.90	.75	.15
10		0	2.96	2.97	.06	2.90	2.56	.21	2.35	2.04	.36	1.68	1.55	.46	1.09	1.22	.51	.71	1.02	.60	.42	.86	.68	.18
1				2.97	.06	2.91	2.56	.20	2.36	2.00	.31	1.69	1.53	.43	1.10	1.28	.55	.73	1.10	.65	.45	.90	.73	.17
2				2.97	.065	2.90	2.58	.22	2.36	2.04	.35	1.69	1.58	.48	1.10	1.32	.61	.71	1.20	.73	.47	1.00	.83	.17
3				2.96	.07	2.89	2.58	.24	2.34	2.05	.38	1.67	1.60	.53	1.07	1.34	.65	.69	1.21	.76	.45	1.03	.87	.16
4				2.96	.09	2.87	2.58	.27	2.31	2.08	.40	1.68	1.62	.55	1.07	1.36	.68	.68	1.21	.77	.44	1.03	.88	.15
5				2.96	.10	2.86	2.58	.27	2.31	2.10	.42	1.68	1.64	.59	1.05	1.36	.68	.68	1.20	.76	.44	1.01	.87	.14
6				2.97	.10	2.87	2.58	.28	2.30	2.14	.46	1.68	1.72	.67	1.05	1.40	.71	.69	1.20	.77	.43	.98	.84	.14
7				2.97	.10	2.87	2.58	.27	2.31	2.15	.47	1.68	1.75	.67	1.08	1.40	.72	.68	1.16	.75	.41	.92	.80	.12
8				2.97	.09	2.88	2.57	.25	2.32	2.14	.46	1.68	1.69	.65	1.04	1.38	.70	.68	1.14	.73	.41	.90	.77	.13
9				2.97	.085	2.87	2.56	.23	2.33	2.10	.42	1.68	1.68	.61	1.07	1.30	.61	.69	1.10	.69	.41	.84	.70	.14
10		0	2.96	2.97	.075	2.88	2.53	.20	2.33	2.00	.34	1.66	1.60	.54	1.06	1.24	.53	.71	1.00	.57	.43	.78	.61	.17

NOTE: ALL PRESSURES IN INCHES OF WATER

DRIVING MOTOR DATA

	MOTOR INPUT		MOTOR	MOTOR INPUT		MOTOR	MOTOR INPUT		MOTOR	MOTOR INPUT		MOTOR	MOTOR INPUT		MOTOR	MOTOR INPUT		MOTOR	MOTOR INPUT		MOTOR	MOTOR INPUT		MOTOR
	VOLTS	AMP.	R.P.M.	VOLTS	AMP.	R.P.M.	VOLTS	AMP.	R.P.M.	VOLTS	AMP.	R.P.M.	VOLTS	AMP.	R.P.M.	VOLTS	AMP.	R.P.M.	VOLTS	AMP.	R.P.M.	VOLTS	AMP.	R.P.M.
1	507	5.5	530	506	7.25	528	506	9.0	523	507	10.0	525	507	10.5	523	505	11.0	518	507	11.5	518	505	11.75	517
2	507	5.6	531	505	7.50	526	505	9.0	527	506	10.0	525	506	10.5	523	505	11.0	517	506	11.5	517	506	11.75	518
3				505	7.50	528	505	9.0	525	505	10.0	525	506	10.5	524	505	11.0	517	505	11.5	517	505	11.75	518

FAN R.P.M.

RUN	1	2	3	4	5	6	7	8
	583	582	574	575	573	568	565	567
	581	579	575	575	572	566	567	568
		578	577	574	572	567	568	568
TEMPERATURES °F								
ROOM	82	80	80	80	80	79	79	78
WET BULB	75	74	74	74	74	73	72	72
DRY BULB	82	80	80	80	80	79	79	78
BAROMETRIC PRESSURE IN. OF Hg.								
	29.75	29.75	29.75	29.75	29.75	29.75	29.75	29.75

Data

The test data are given on page 19.

Method of Calculation

Standard air for these calculations is taken as dry air at 70° F., barometer 29.92 in. of mercury, which corresponds to a density of .07495 lb. per cu. ft.

All data are corrected to a constant speed of 582 r.p.m.

Referring to the original data of Run No. 3, the average static pressure for both traverses will be

2.33 in. of water = 2.33/13.6 = .171 in. of Mercury.

The average velocity pressure is computed according to equation (6). This average velocity pressure is

.242 in. of water.

The air pressure in the duct is equal to the barometric pressure plus the static pressure in inches of mercury.

29.75 + .171 = 29.92 in. of mercury

Dry bulb temperature = 80°F

Wet bulb temperature = 74°F

Air density at the above pressure and temperatures .0728 lb. per cu. ft.

The input horsepower to the fan is taken as the output horsepower of the driving motor decreased by two per cent for losses in the belt transmission. The horsepower output of the driving motor for each run is given in the data on the calibration of the driving motor which is given in Chapter VI. Thus,

Horsepower output = 3.4

Horsepower input to fan = .98 x 3.4 = 3.33

In order to reduce the results to standard air the static pressure, velocity pressure, and horsepower have to be multiplied by the ratio of the density of standard air to that of the test air, as explained on page 13. Thus,

Static pressure = 2.33 x .07495/.0728 = 2.40 in.

Velocity pressure = .242 x .07495/.0728

= .249

Horsepower = 3.33 x .07495/.0728 = 3.43

In order to reduce the results to a constant speed of 582 r.p.m. it is necessary to multiply the pressures by the square of the speed ratio and the horsepower by the cube of the speed ratio, as explained on page 11. Thus,

Speed ratio = 582/575.3 = 1.113

$1.013^2 = 1.027$

and

$1.013^3 = 1.04$

Therefore Static pressure = 2.4 x 1.027 = 2.46

Velocity pressure = .249 x 1.027 = .255

Horsepower = 3.43 x 1.04 = 3.57

In order to correct for the loss in static pressure in the discharge duct, equation (7) is applied;

f = .0257 L/D VP

f = .0257 x 13.75/1.833 x .255 = .0492 in. of water

The total pressure at the fan outlet will be the sum of the static pressure at the fan outlet and the velocity pressure, according to equation (3).

Static pressure at fan = 2.46 + .049 = 2.509 in.

Total pressure at fan = 2.509 + .255 = 2.764 in.

In order to determine the velocity in the duct equation (5) is applied.

$$V = 1096.2 \sqrt{.255/.07495} = 2025 \text{ ft. per min.}$$

The capacity is equal to the velocity times the area of the duct.

$$\text{Duct area} = \pi/4 \times (22/12)^2 = 2.64 \text{ sq. ft.}$$

Capacity = 2025 x 2.64 = 5330 cu. ft. per min.

The horsepower transmitted to the air by the fan is according to equation (4)

$$AHP = \frac{5330 \times 2.764}{6356}$$

$$= 2.32$$

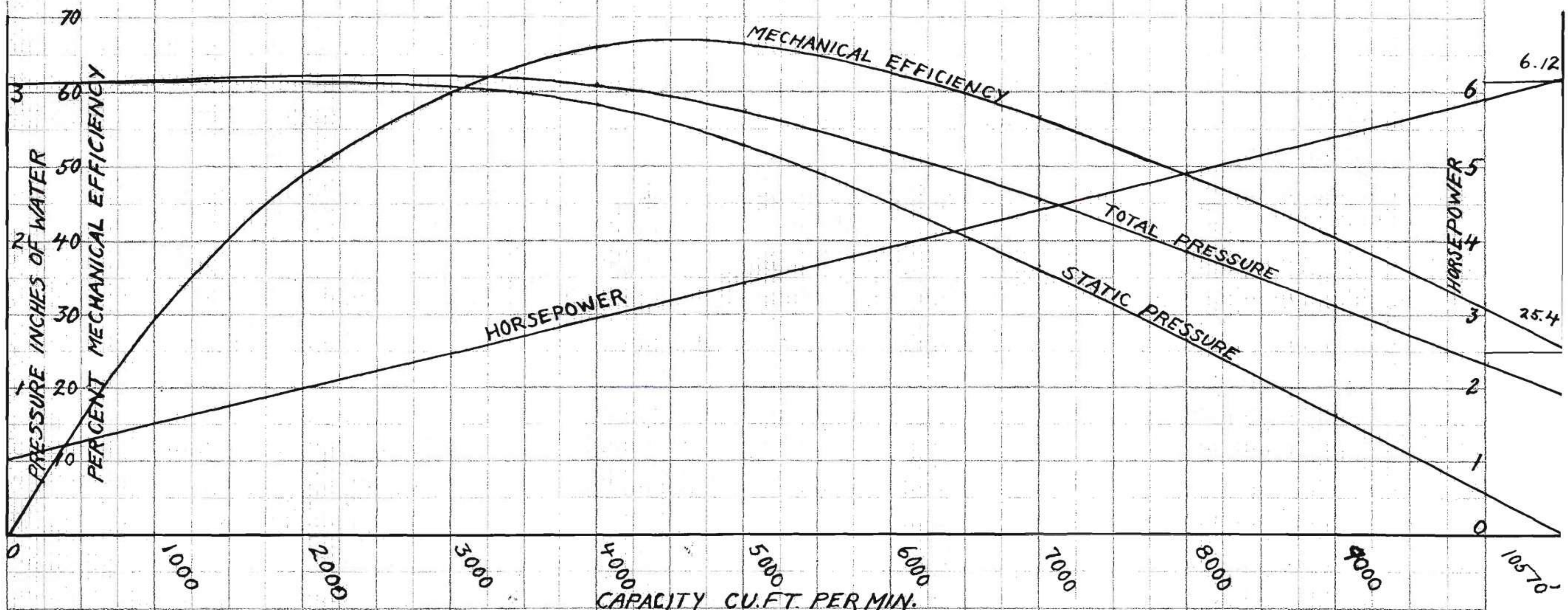
The mechanical efficiency is equal to the ratio of the air horsepower to the input horsepower. Thus,

Mechanical Efficiency = 2.32/3.57 = 65 per cent.

CONDENSED SUMMARY OF TEST DATA

Corrected to Standard Air 70°F dry, barometric pressure 29.92 in. of mercury; corrected to a constant speed of 582 r.p.m.

Run No.	Pressures Total in. of water	Static in. of water	Capacity Cu. ft. per min.	Air Horse-power	Input Horse-power	Mechanical Efficiency per cent
1	3.05	3.05	0	0	1.05	0
2	3.09	3.01	3090	1.55	2.44	61.6
3	2.76	2.51	5330	2.32	3.57	65.0
4	2.28	1.86	6870	2.47	4.26	57.8
5	1.87	1.27	8230	2.43	4.71	51.5
6	1.60	0.90	8860	2.23	5.16	43.2
7	1.42	0.63	9410	2.11	5.61	37.6
8	1.21	0.34	9860	1.88	5.76	32.6



RESULTS OF TEST ON NO.9 FAN AT 582 R.P.M.

Conclusions

A plot of the test results as given on page 24 shows the following to be true:

1. The fan has a very good mechanical efficiency.
2. The horsepower required to drive the fan varies directly as the capacity for a given speed.
3. For the working range of the fan, i.e., for a capacity corresponding to the point of maximum mechanical efficiency and for capacities above this point, the static pressure has a rising value, for a decrease in capacity. This fact will allow this fan to operate satisfactorily in parallel with another fan because if it starts to pump less than it should its pressure would have to rise and this would make it pump more when in parallel with other fans.

III. GENERAL CHARACTERISTIC CURVES

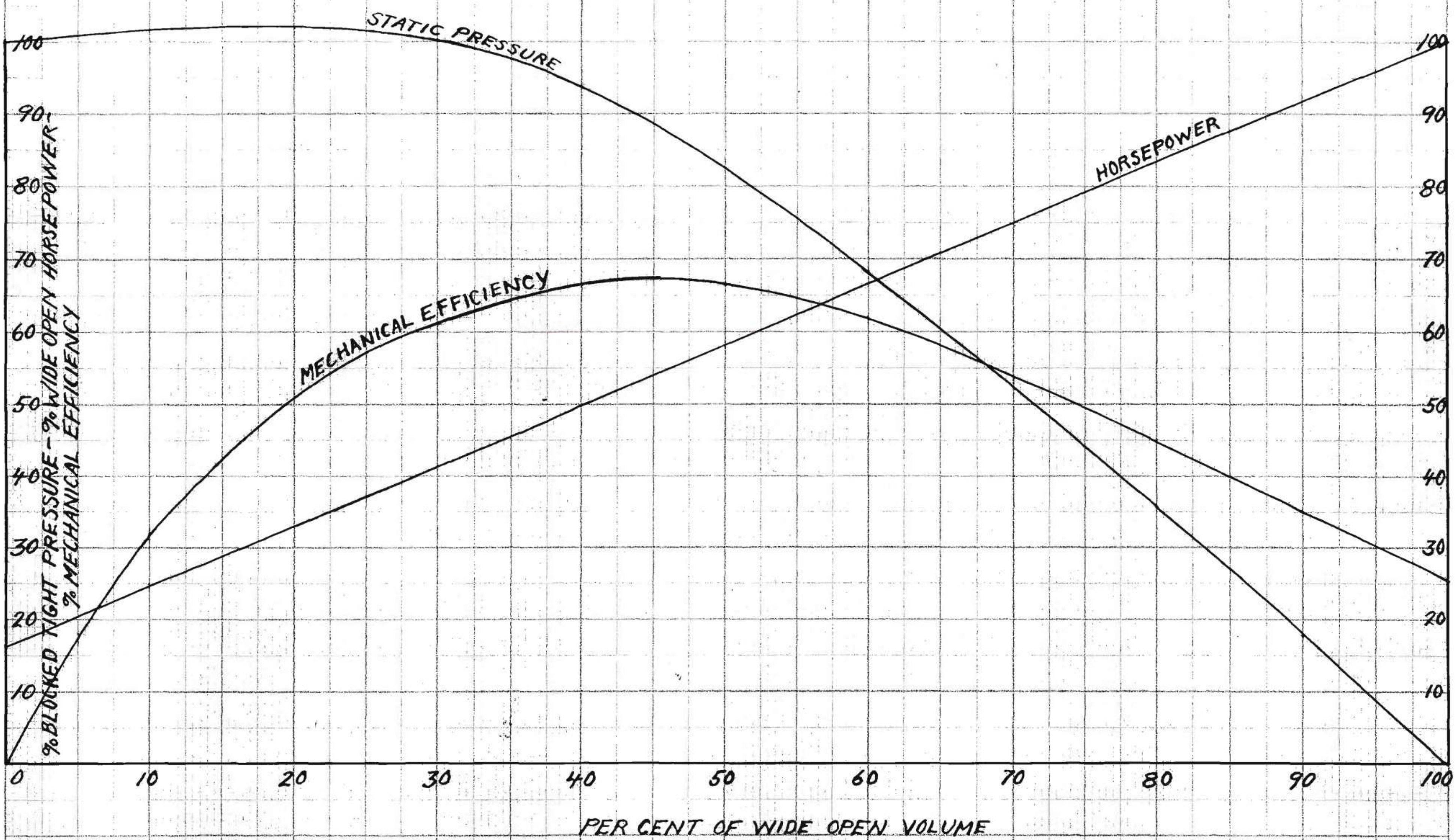
The general operating characteristics of these Fans are given in graphical form on page 27. These curves show the variation in mechanical efficiency, static pressure, and horsepower, with variation in per cent of wide open capacity. These curves are applicable to any size fan driven at any constant speed.

The data for these curves was computed from values read from the test curves on page 24.

As an illustration of one of the uses of these curves, the following example is given:

A number nine fan is to be driven at 615 r.p.m. and is to deliver 6,000 cu. ft. per min. against a static pressure of 2.66 inches of water and it is desired to know what the mechanical efficiency will be at this rating.

Solution: The performance chart for a No. 9 fan which is given in chapter V shows that the fan will deliver 11,150 cu. ft. per min. when wide open for this speed. The point of rating then is 6,000/11,150 or 53.8 per cent of wide open volume. The general characteristic curve shows that the fan will have a mechanical efficiency of 64.7 per cent at this rating.



GENERAL CHARACTERISTIC CURVES

IV. CAPACITY TABLES

Laws Upon Which the Tables are Based

A given fan which is operating at a definite capacity, pressure, and horsepower will have a definite mechanical efficiency, as can be seen from the general characteristic curves on page 27. If the speed is changed and the outlet conditions kept the same as before, the mechanical efficiency will remain the same, but the capacity, pressure, and horsepower will change according to the laws given on page 11.

1. Capacity varies directly as the speed ratio.
2. Pressure varies as the square of the speed ratio.
3. Horsepower varies as the cube of the speed ratio.

For a series of different size fans which are symetrical in design and which are operated at the same peripheral speed, the following laws apply, which are given on page 13.

1. Speed varies inversely as the fan size.
2. Capacity varies as the square of the fan size.
3. Horsepower varies as the square of the fan size.
4. Pressure will be the same for each size.

Relative Sizes of the Fans.

The following dimensions will illustrate the relative sizes of the Fans;

Size No.	Impeller Diam. in.	Width in.	Inlet & Outlet in.	Relative Size
7	25	9.25	15	.625
8	32	12	18	.800
9	40	15	22	1.00
10	50	19	28	1.25
11	60	23	33	1.50
12	70	26	38	1.75

Method of Calculation

The test results on the No. 9 fan show that the machine will have an efficiency of 67 per cent against a static pressure of 2.69 in. of water and that 3.30 horsepower will be required. Now, if the machine is operated at 795 r.p.m. the speed ratio will be

795/582 = 1.364

The quantities at 795 r.p.m. will be:

Capacity = 1.364 x 4750 = 6470 cu. ft. per min.

Static pressure = 1.364^2 x 2.69 = 5.00 in. of water.

Horsepower = 1.364^3 x 3.30 = 8.37

Mechanical efficiency = 67 per cent.

Tip speed = π x 40/12 x 795 = 8330 ft. per min.

Suppose a No. 10 fan is to be operated the same static pressure and efficiency, and it is desired to know what the other quantities will be.

Relative size = 1.25

Static pressure = 5.00 in. of water.

Mechanical efficiency = 67 per cent.

Speed = 795/1.25 = 636 r.p.m.

Capacity = $6470 \times 1.25^2 = 10{,}100$ cu. ft. per min.

Horsepower = $8.37 \times 1.25^2 = 13.1$

PERFORMANCE OF NO.7 FAN

FOR DRY AIR 70°F. BAROMETER 29.92 IN.

R.P.M.	TIP SPEED FT. PER MIN.	S.P. 1/2" WATER .288 OZ.		S.P. 1" WATER .577 OZ.		S.P. 1 1/2" WATER .865 OZ.		S.P. 2" WATER 1.153 OZ.		S.P. 2 1/2" WATER 1.442 OZ.		S.P. 3" WATER 1.730 OZ.		S.P. 3 1/2" WATER 2.018 OZ.		S.P. 4" WATER 2.307 OZ.		S.P. 5" WATER 2.884 OZ.		S.P. 6" WATER 3.460 OZ.		S.P. 7" WATER 4.007 OZ.	
		C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.
568	3720	1900	.4	1130	.3																		
695	4550	2600	.85	2100	.7	1390	.54																
804	5270	3130	1.4	2650	1.2	2200	1.0	1600	.83														
898	5870	3600	1.9	3200	1.8	2750	1.6	2300	1.4	1790	1.2												
984	6440			3700	2.4	3300	2.3	2900	2.0	2450	1.8	1960	1.5										
1062	6950			4050	3.1	3700	2.9	3350	2.7	3000	2.5	2600	2.2	2120	1.9								
1137	7430					4100	3.7	3800	3.4	3450	3.2	3100	2.9	2700	2.6	2260	2.34						
1271	8330					4800	5.3	4500	5.0	4200	4.7	3900	4.5	3600	4.2	3300	4.0	2530	3.3				
1390	9100							5150	6.9	4900	6.6	4600	6.3	4350	6.0	4070	5.7	3500	5.1	2770	4.3		
1530	10000							5900	9.4	5650	9.1	5400	8.8	5200	8.5	4930	8.2	4430	7.5	3870	6.8	3200	5.9

PERFORMANCE OF NO. 8 FAN

FOR DRY AIR 70°F. BAROMETER 29.92 IN.

R.P.M.	TIP SPEED FT. PER MIN.	S.P. ½" WATER .288 OZ.		S.P. 1" WATER .577 OZ.		S.P. 1½" WATER .865 OZ.		S.P. 2" WATER 1.153 OZ.		S.P. 2½" WATER 1.442 OZ.		S.P. 3" WATER 1.730 OZ.		S.P. 3½" WATER 2.018 OZ.		S.P. 4" WATER 2.307 OZ.		S.P. 5" WATER 2.884 OZ.		S.P. 6" WATER 3.460 OZ.		S.P. 7" WATER 4.007 OZ.	
		C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.
444	3720	3100	.7	1860	.48																		
543	4550	4230	1.4	3300	1.1	2270	.88																
628	5270	5140	2.2	4400	2.0	3600	1.7	2620	1.4														
702	5870	5900	3.3	5200	2.9	4500	2.6	3800	2.2	2930	1.9												
768	6440			5950	4.0	5350	3.6	4700	3.3	4000	2.9	3220	2.5										
828	6950			6600	5.1	6000	4.7	5450	4.3	4900	4.0	4250	3.6	3470	3.1								
887	7430					6700	6.0	6200	5.6	5650	5.2	5100	4.8	4500	4.4	3700	3.8						
993	8330					7900	8.7	7400	8.2	7000	7.8	6500	7.4	6000	7.0	5400	6.5	4140	5.4				
1086	9100							8400	11.2	8000	10.8	7600	10.3	7100	9.8	6700	9.4	5700	8.4	4530	7.0		
1193	10000							9600	15.2	9200	14.8	8800	14.3	8400	13.7	8000	13.2	7200	12.2	6360	11.0	5300	9.7

PERFORMANCE OF NO. 9 FAN

FOR DRY AIR 70°F BAROMETER 29.92 IN.

R.P.M.	TIP SPEED FT. PER MIN.	S.P. ½" WATER .288 OZ.		S.P. 1" WATER .577 OZ.		S.P. 1½" WATER .865 OZ.		S.P. 2" WATER 1.153 OZ.		S.P. 2½" WATER 1.442 OZ.		S.P. 3" WATER 1.730 OZ.		S.P. 3½" WATER 2.018 OZ.		S.P. 4" WATER 2.307 OZ.		S.P. 5" WATER 2.884 OZ.		S.P. 6" WATER 3.460 OZ.		S.P. 7" WATER 4.007 OZ.	
		C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.
355	3720	5000	1.1	2900	.75																		
434	4550	6600	2.2	5300	1.75	3550	1.4																
502	5270	8050	3.6	6900	3.2	5700	2.6	4100	2.1														
561	5870	9300	5	8200	4.95	7100	4.0	5900	3.6	4600	3.0												
615	6440			9350	6.3	8400	5.7	7450	5.2	6350	4.7	5020	3.9										
663	6950			10300	8.0	9400	7.4	8550	6.8	7600	6.3	6600	5.6	5400	4.9								
710	7430					10500	9.4	9700	8.8	8850	8.3	8000	7.5	7000	7.0	5800	6.0						
795	8330					12300	13.6	11600	13.0	10800	12.8	10050	11.6	9300	11.0	8450	10.3	6470	8.4				
869	9100							13200	17.7	12500	17.0	11900	16.3	11200	15.5	10450	14.8	8900	13.0	7080	12.0		
955	10000							15000	24.0	14400	23.3	13800	22.5	13200	21.7	12600	20.9	11300	19.3	9900	17.4	8300	15.3

PERFORMANCE OF NO. 10 FAN

FOR DRY AIR 70°F BAROMETER 29.92 IN.

R.P.M.	TIP SPEED FT. PER MIN.	S.P. ½" WATER .288 OZ.		S.P. 1" WATER .577 OZ.		S.P. 1½" WATER .865 OZ.		S.P. 2" WATER 1.153 OZ.		S.P. 2½" WATER 1.442 OZ.		S.P. 3" WATER 1.730 OZ.		S.P. 3½" WATER 2.018 OZ.		S.P. 4" WATER 2.307 OZ.		S.P. 5" WATER 2.884 OZ.		S.P. 6" WATER 3.460 OZ.		S.P. 7" WATER 4.007 OZ.	
		C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.
284	3720	7500	1.7	4500	1.2																		
348	4550	10300	3.3	8200	2.9	5500	2.1																
402	5270	12500	5.5	10700	5.0	8800	4.3	6400	3.4														
449	5870	14300	7.8	12700	7.0	11100	6.3	9200	5.6	7200	4.6												
492	6440			14600	9.8	13100	9.0	11600	8.1	9900	7.2	7800	6.1										
531	6950			16100	12.4	14800	11.6	13300	10.7	12000	10.0	10400	9.0	8400	7.6								
568	7430					16400	14.7	15200	13.8	13800	12.9	12400	11.8	10900	10.7	9000	9.4						
636	8330					19200	21.3	18000	20.2	16900	19.5	15800	18.3	14400	17.2	13100	16.0	10100	13.1				
695	9100							20600	27.5	19600	26.5	18600	25.3	17500	24	16400	23	14000	20.4	11000	17.2		
764	10000							23500	38.0	22500	36.5	21600	35.2	20600	34	19600	32.8	17600	30	15400	27.2	12800	23.8

PERFORMANCE OF NO. 11 FAN

FOR DRY AIR 70°F BAROMETER 29.92 IN.

R.P.M.	TIP SPEED FT. PER MIN.	S.P. ½" WATER .288 OZ.		S.P. 1" WATER .577 OZ.		S.P. 1½" WATER .865 OZ.		S.P. 2" WATER 1.153 OZ.		S.P. 2½" WATER 1.442 OZ.		S.P. 3" WATER 1.730 OZ.		S.P. 3½" WATER 2.018 OZ.		S.P. 4" WATER 2.307 OZ.		S.P. 5" WATER 2.884 OZ.		S.P. 6" WATER 3.460 OZ.		S.P. 7" WATER 4.007 OZ.	
		C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.
237	3720	11000	2.4	6500	1.7																		
289	4550	14900	5.0	11900	4.1	8000	3.1																
335	5270	18000	8.0	15500	7.0	12800	6.0	9200	4.8														
374	5870	20700	11.5	18400	10.4	16000	9.3	13400	8.1	10300	6.6												
410	6440			21100	14	19000	13	16800	118	14300	103	11300	8.8										
441	6950			23200	18	21300	16.8	19300	15.4	17300	14.2	15000	13.9	12200	11								
473	7430					23600	21	21800	19.9	19900	18.3	17900	17	15700	15.5	13000	13.5						
529	8330					27600	30.5	26000	29	24300	275	22600	26	20800	24.2	19000	225	14600	188				
579	9100							27700	39.5	28200	38	26600	363	25100	34.6	23500	33	20000	29	16000	248		
636	10000							33800	54.4	32500	525	31100	50.6	29700	49	28200	47	25400	43	22200	39	18600	34

PERFORMANCE OF NO. 12 FAN

FOR DRY AIR 70°F BAROMETER 29.92 IN.

R.P.M.	TIP SPEED FT. PER MIN.	S.P. ½" WATER .288 OZ.		S.P. 1" WATER .577 OZ.		S.P. 1½" WATER .865 OZ.		S.P. 2" WATER 1.153 OZ.		S.P. 2½" WATER 1.442 OZ.		S.P. 3" WATER 1.730 OZ.		S.P. 3½" WATER 2.018 OZ.		S.P. 4" WATER 2.307 OZ.		S.P. 5" WATER 2.884 OZ.		S.P. 6" WATER 3.460 OZ.		S.P. 7" WATER 4.007 OZ.	
		C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.	C.F.M.	H.P.
203	3720	15100	3.7	9000	2.3																		
248	4550	20300	6.9	16100	5.8	10900	4.2																
287	5270	24500	10.9	21000	9.5	17100	8.1	12600	6.5														
321	5870	28300	15.5	25100	14	21900	12.6	17100	11	14000	9												
351	6440			28400	19.2	25800	17.8	22500	16	19300	14	15400	11.9										
379	6950			31800	24.3	29000	22.5	26200	21	23700	19.1	20300	17.2	16600	14.9								
406	7430					32000	29	29300	27	26600	25.2	24000	23	21000	21	17700	18.4						
454	8330					37800	41.2	35400	39.5	33100	37.5	30900	35.4	28300	33.3	25900	31	19800	25.6				
497	9100							40300	54	38300	51.8	36100	49.4	34000	47	32000	45	27300	39.8	21700	33.7		
546	10000							46300	73	44300	70.6	42500	68	40500	65.7	38500	63	34500	58	30100	52.8	25000	46.4

V. PERFORMANCE CHARTS

The following six pages contain the performance charts for all size fans. These charts have the advantage over the capacity tables in that they present all significant operating data within the operating range of the machine.

The charts are plotted from data which is calculated in the same manner as was explained in connection with the capacity tables.

These charts are original in the method of presenting the data and are unique in that no calculations are required to determine the operating characteristics of any size fan.

Use of the Charts

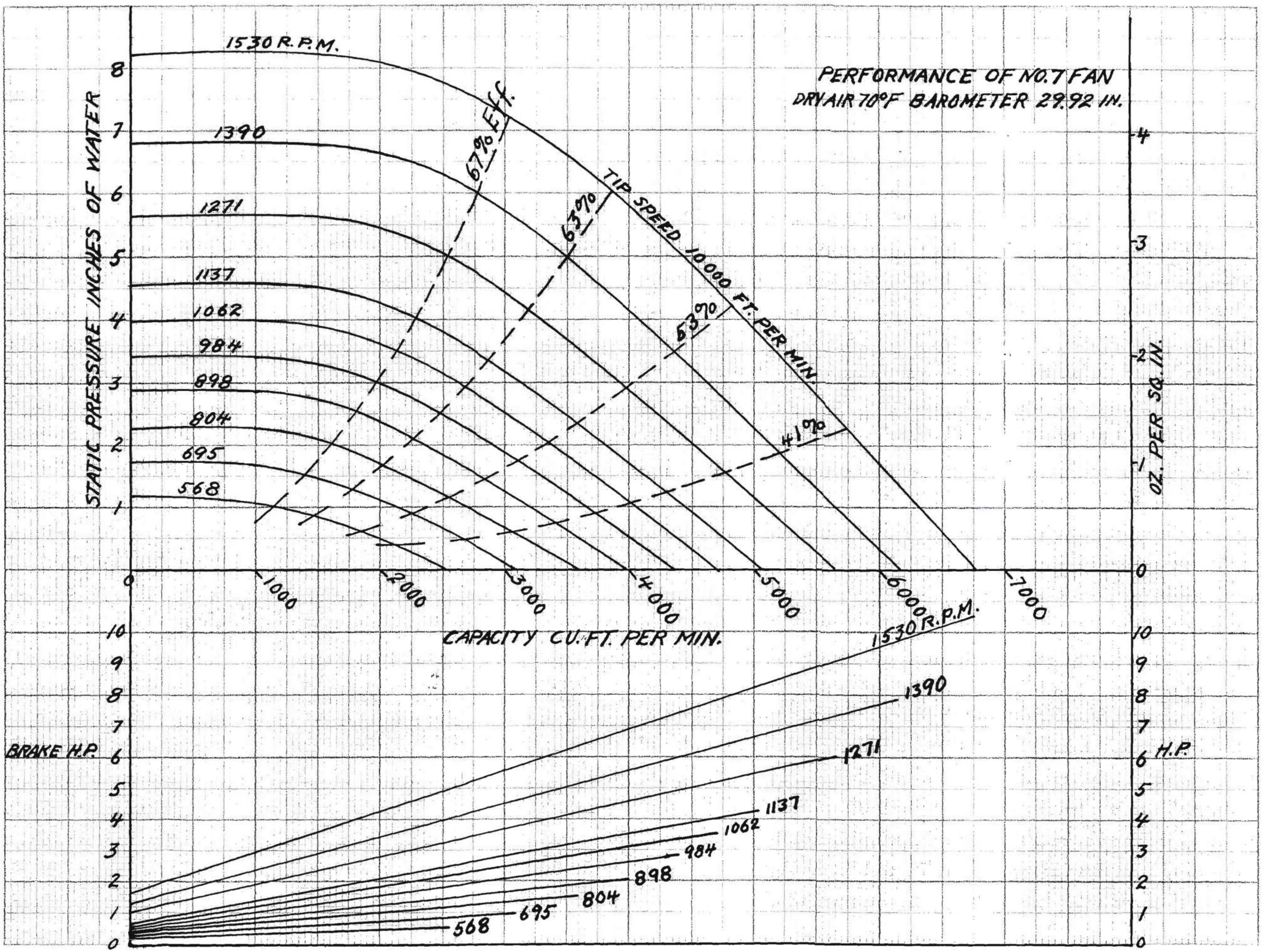
The following examples will illustrate some of the uses of the charts.

1. It is desired to handle 9,200 cu. ft. per min. of standard air against a static pressure of 2 in. of water.

   The charts show that it is possible to handle this amount with all sizes except No. 7. The No. 11 chart shows that this machine will give the required performance at 335 r.p.m. and will require 4.8 horsepower. The No. 10 chart shows that this machine will give the required performance when driven at 449 r.p.m. but will require 5.6 horsepower. The mechanical efficiency in the first case will be 67 per cent and in the second case 62 per cent.

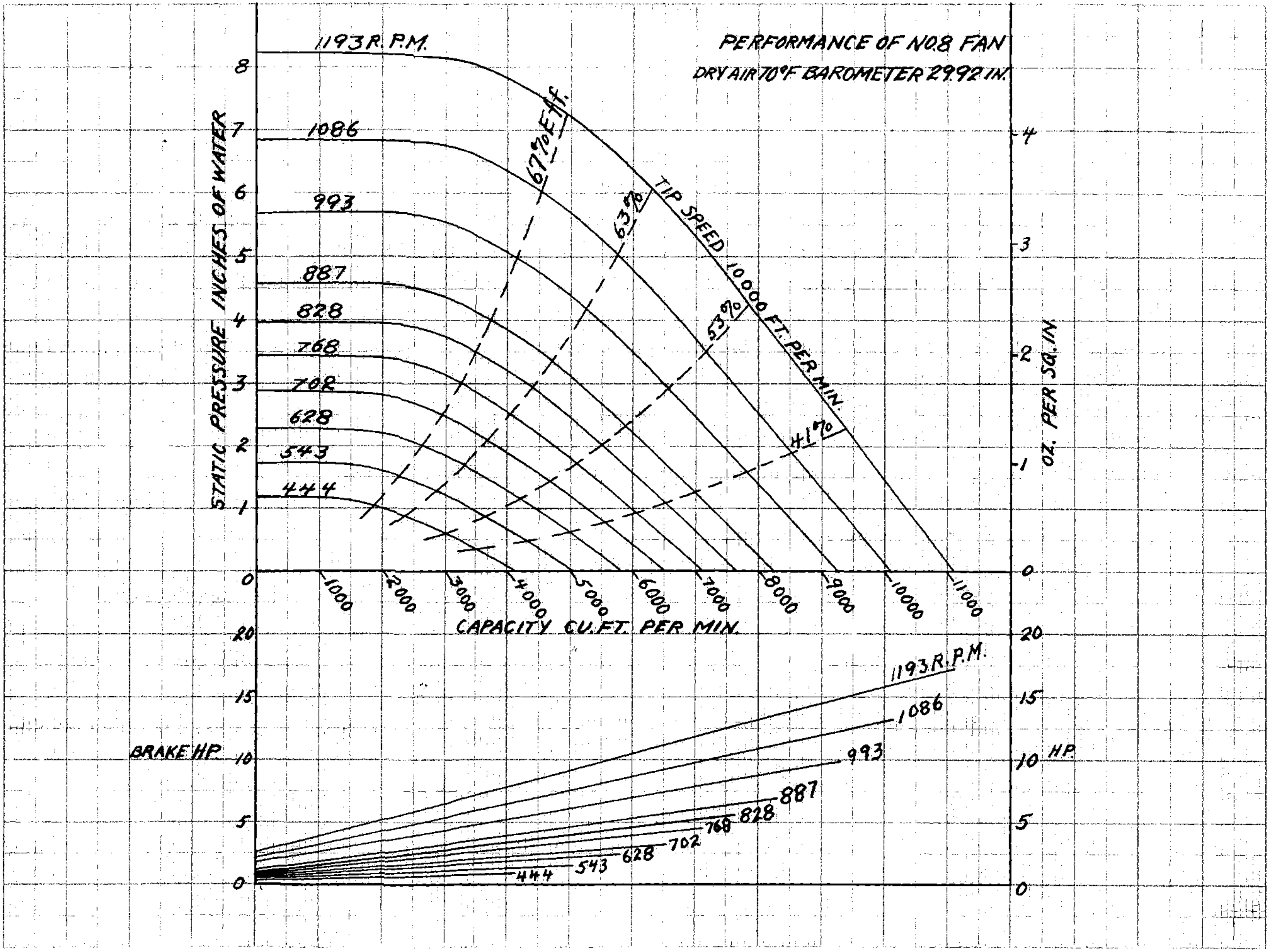
2. Suppose that a No. 11 machine was supplied in the above case and that later the pressure was changed to 1-1/2 in. of water. What would the new capacity and horsepower change to, and what value would the efficiency have?

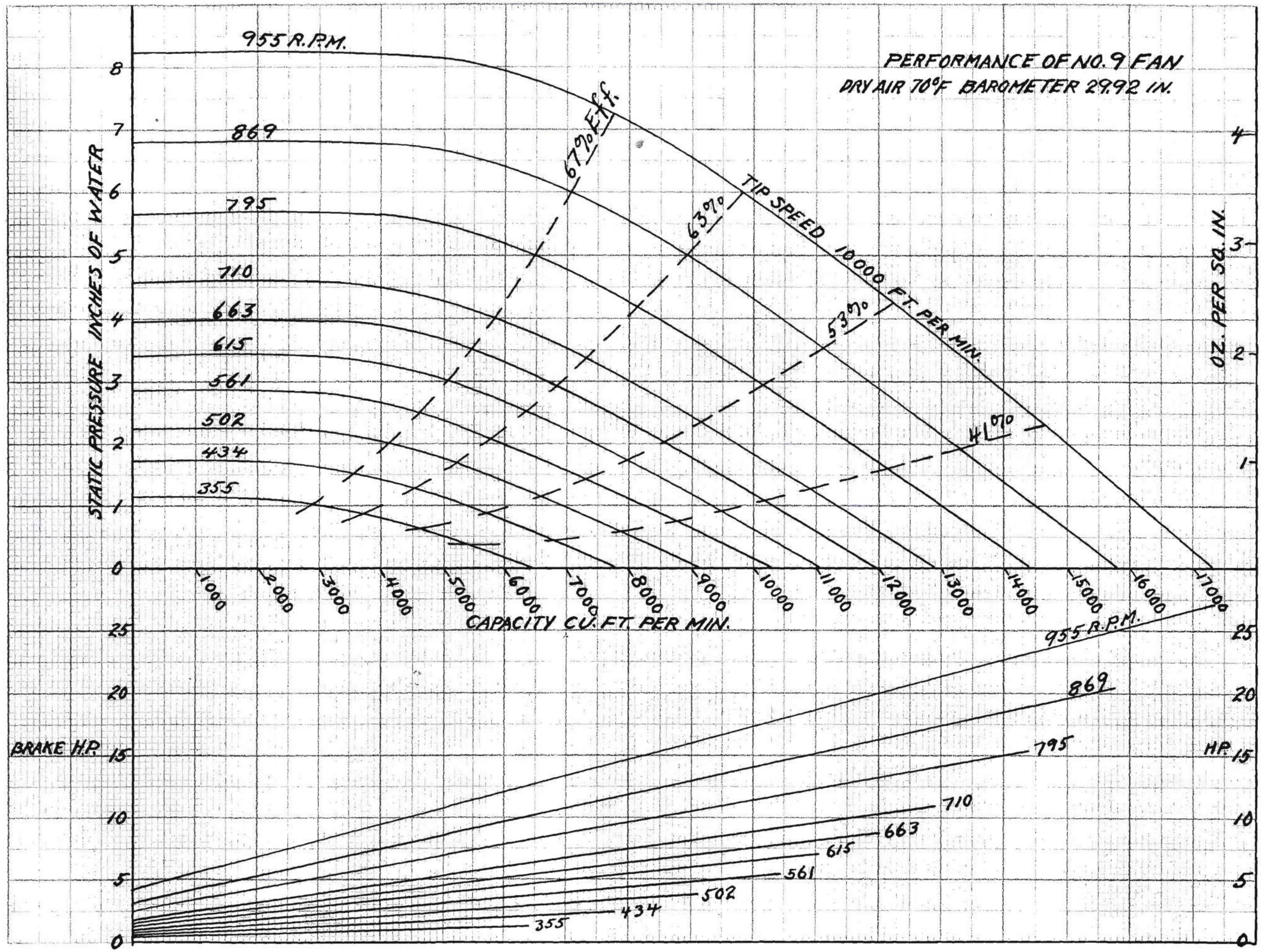
The No. 11 chart shows that at 335 r.p.m. and 1-1/2 in. pressure, the capacity will be 12,800 cu. ft. per min., the horsepower will be 6.0, and the efficiency will be 60 per cent.



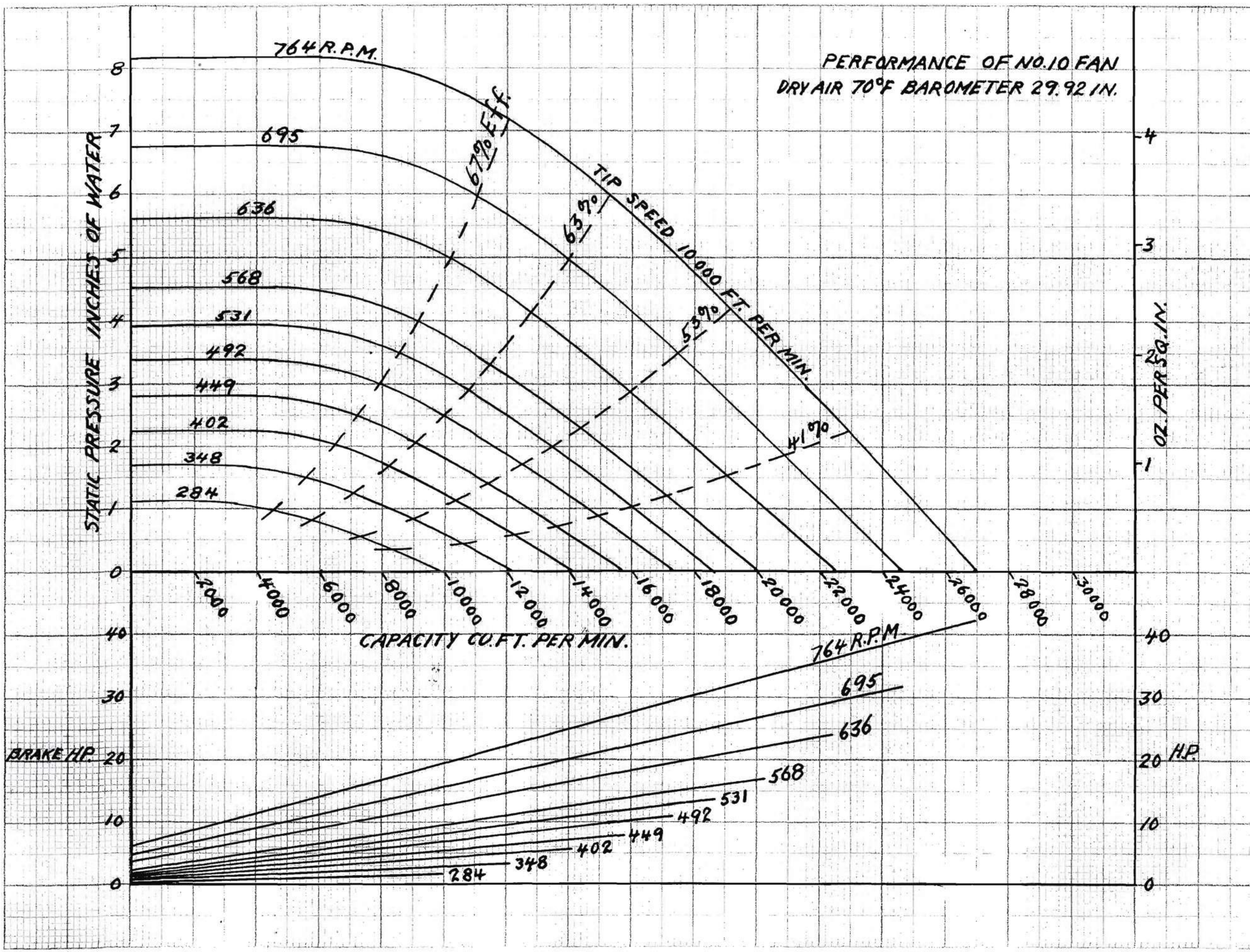
PERFORMANCE OF NO.7 FAN
DRY AIR 70°F BAROMETER 29.92 IN.
STATIC PRESSURE INCHES OF WATER
OZ. PER SQ. IN.
CAPACITY CU.FT. PER MIN.
BRAKE H.P.
H.P.
TIP SPEED 10 000 FT. PER MIN.
67% Eff.
63%
53%
41%
1530 R.P.M.
1390
1271
1137
1062
984
898
804
695
568
1000
2000
3000
4000
5000
6000
7000

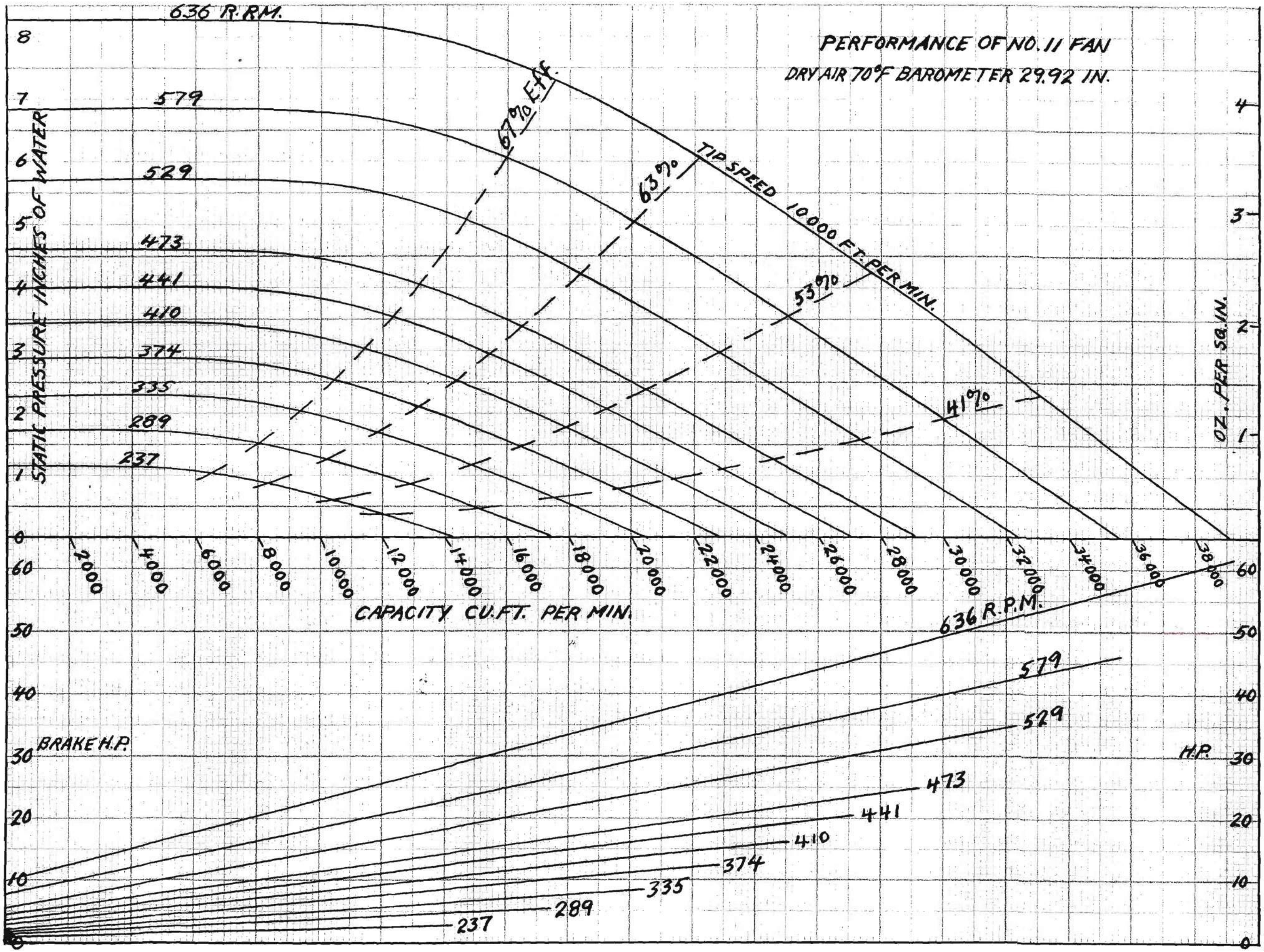
39


PERFORMANCE OF NO.8 FAN
DRY AIR 70°F BAROMETER 29.92 IN.
1193 R.P.M.
1086
993
887
828
768
702
628
543
444
67% Eff.
63%
53%
41%
TIP SPEED 10000 FT. PER MIN.
STATIC PRESSURE INCHES OF WATER
OZ. PER SQ. IN.
CAPACITY CU.FT. PER MIN.
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
11000
BRAKE HP.
HP.
0
1
2
3
4
5
6
7
8
10
15
20


PERFORMANCE OF NO. 9 FAN
DRY AIR 70°F BAROMETER 29.92 IN.
STATIC PRESSURE INCHES OF WATER
OZ. PER SQ. IN.
955 R.P.M.
869
795
710
663
615
561
502
434
355
67% Eff.
63%
53%
41%
TIP SPEED 10000 FT. PER MIN.
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
11000
12000
13000
14000
15000
16000
17000
CAPACITY CU. FT. PER MIN.
BRAKE H.P.
H.P.

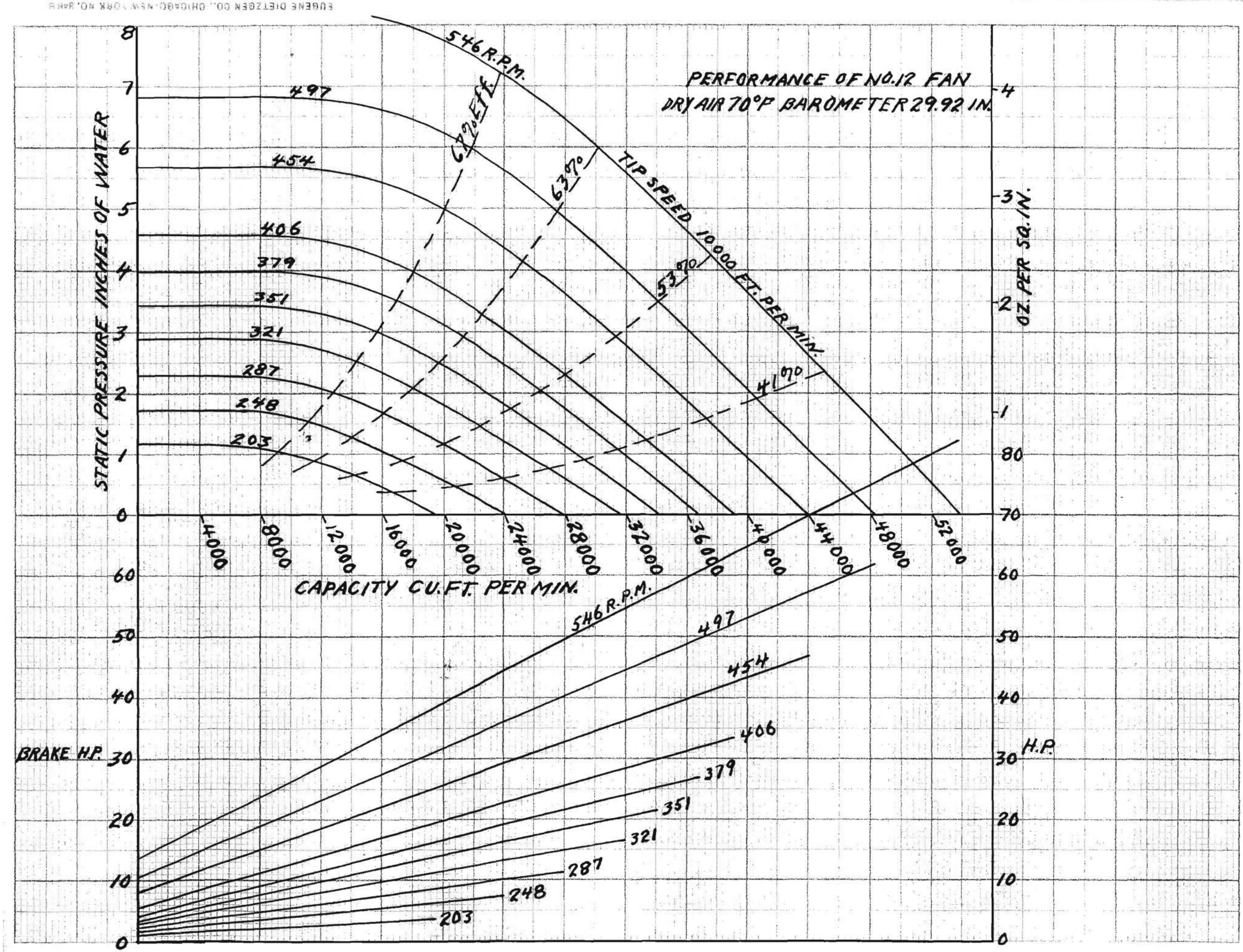
EUGENE DIETZGEN CO., CHICAGO-NEW YORK NO. 346 R





PERFORMANCE OF NO. 11 FAN
DRY AIR 70°F BAROMETER 29.92 IN.
636 R.P.M.
579
529
473
441
410
374
335
289
237
67% Eff.
63%
53%
41%
TIP SPEED 10000 FT. PER MIN.
STATIC PRESSURE INCHES OF WATER
OZ. PER SQ. IN.
CAPACITY CU.FT. PER MIN.
2000
4000
6000
8000
10000
12000
14000
16000
18000
20000
22000
24000
26000
28000
30000
32000
34000
36000
38000
BRAKE H.P.
H.P.

EUGENE DIETZGEN CO., CHICAGO-NEW YORK



VI. CALIBRATION OF DRIVING MOTOR

Data on Motor

General Electric Company, Direct Current Motor,

Serial Number 616284,

Shunt Wound, Form A, RLC, 203B-4,

25 H. P., 400-1200 R.P.M.

40 Amp., 550 Volts.

Note: Motor has been rewound.

Method of Calibration

The motor was calibrated by determining the various losses and subtracting these from the input energy. The losses were determined for the same field current and speed as was used in the fan test.

The field resistance was measured with a Wheatstone Bridge after the motor was up to temperature. The armature resistance was measured for five different positions, readings being taken with the Wheatstone Bridge.

All voltage and ampere readings were corrected according to the calibration curve given at the end of this chapter.

Data (averages)

R.P.M.	530
Field resistance	518.6 ohms
Armature resistance	1.168 ohms
Total input amp. (no load)	4.2
Voltage	505

Total input watts (no load) - - - - - - 2120

Field amp. - - - - - - - - - - - - - - - .973

Armature amp. (no load) - - - - - - - - 3.227

Field loss - - - - - - - - - - - - - - 491 watts

Armature loss (no load) - - - - - - - - 12 watts

Stray power - - - - - - - - - - - - - - 1617 watts

DATA FOR FAN TEST

Run No.	Motor r.p.m.	Volts	True Amp.	K.W. Input	K.W. Loss	K.W. Output	HP. Output	Motor Efficiency
1	530.5	507.0	5.75	2.914	2.135	0.779	1.04	26.8
2	527.3	505.3	7.75	3.92	2.162	1.758	2.35	44.8
3	525.0	505.3	9.36	4.73	2.190	2.540	3.40	53.7
4	525.0	506.0	10.36	5.24	2.211	3.029	4.06	57.8
5	523.3	506.3	10.90	5.52	2.223	3.297	4.42	59.8
6	517.3	505.0	11.40	5.75	2.235	3.515	4.71	61.2
7	517.3	506.0	11.92	6.03	2.248	3.782	5.08	62.8
8	517.7	505.3	12.20	6.17	2.255	3.915	5.25	63.5

Typical Calculation

Run No. 1

K. W. Input = 507.0 x 5.75 x .001 = 2.914

K. W. Loss = Stray power + Field loss + Armature loss

$$= (1617 + 491 + (5.75 - .973)^2 \times 1.168) \times .001$$

$$= 2.135$$

K. W. Output = 2.914 - 2.135 = 0.779

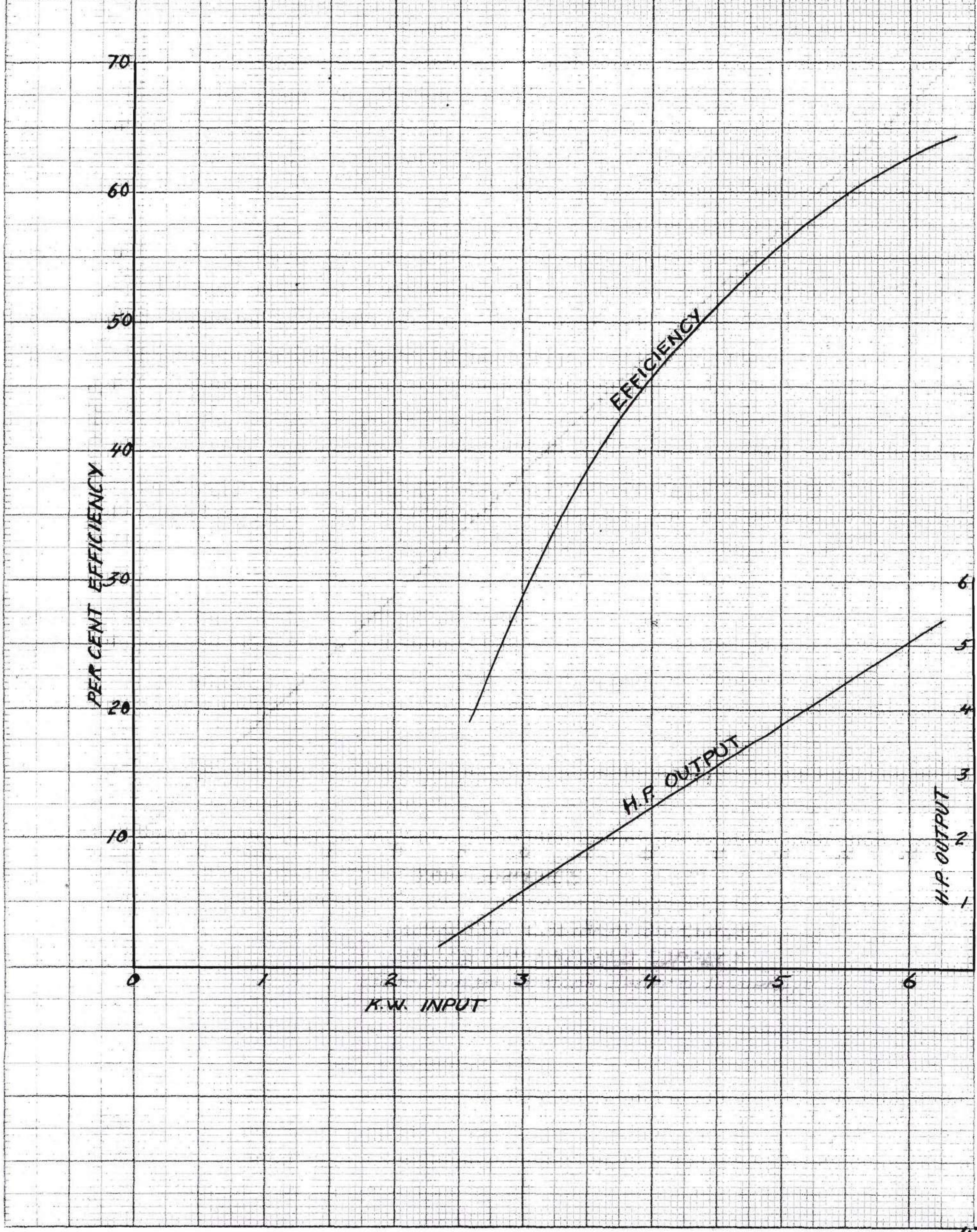
Horsepower Output = 0.779/.746 = 1.04

Motor Efficiency = 0.779/2.914 = 26.8

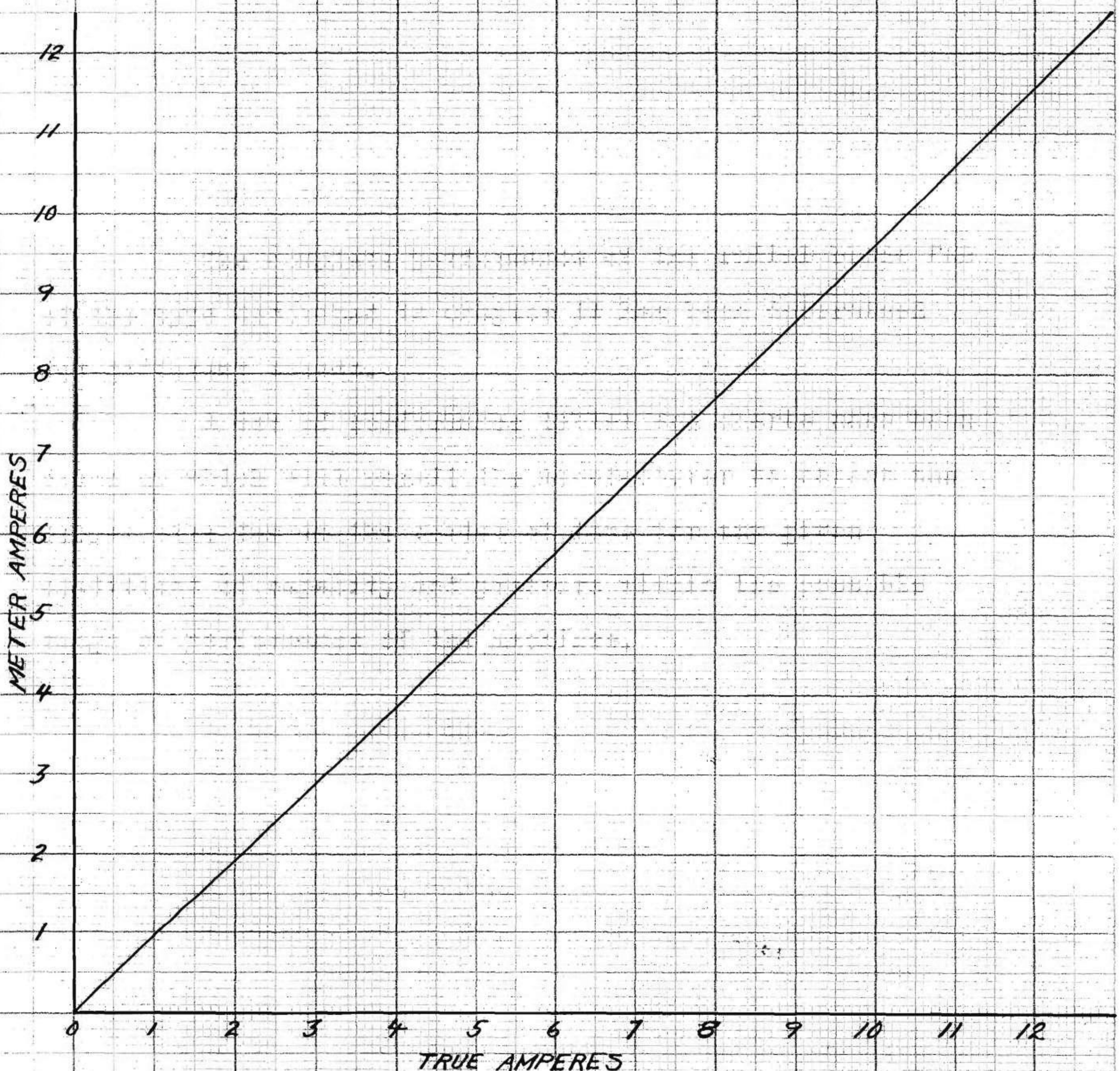
Curves

The efficiency and horsepower output curves are given on page 47.

The ammeter and voltmeter calibrations are given on page 48.



EUGENE DIETZGEN CO. CHICAGO NEW YORK NO. 340 B



AMMETER CALIBRATION CURVE
VOLTMETER CHECKED CORRECT
(Data Furnished by B'ham Electric Co. Laboratory)

EUGENE DIETZGEN CO., CHICAGO-NEW YORK NO. 346

CONCLUSIONS

The complete performance of the radial blade fan of the type described in Chapter II has been determined for different speeds.

A set of performance tables and charts have been drawn up which will permit the manufacturer to select the proper size fan in the series of fans for any given conditions of capacity and pressure within the possible range of performance of the machines.

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